

Normally-On Drum Brake with External Shoes and an Electrohydraulic Thrustor.

§ Given data:

* Thrustor:

- Max. force/max. stroke 300 N/25 mm
- Actuation time 0.2 s
- Motor Power 110 W
- Pressure 60 kPa
- Bore/max. outside Diam. 90 mm/120 mm
- Closed height \approx 300 mm

* Actuating Lever:

- a/b/c = 22.5/45/135 mm

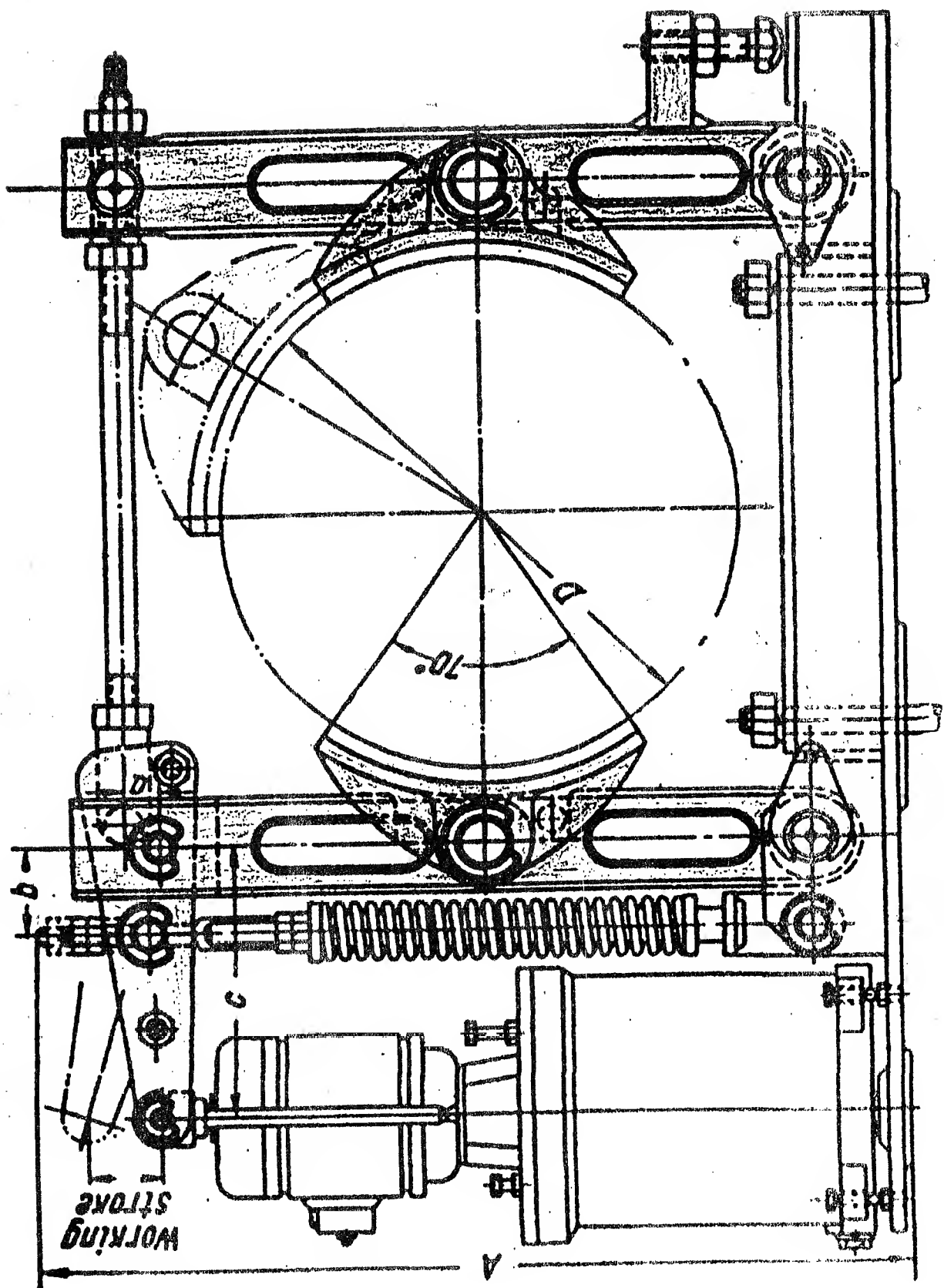
* Brake:

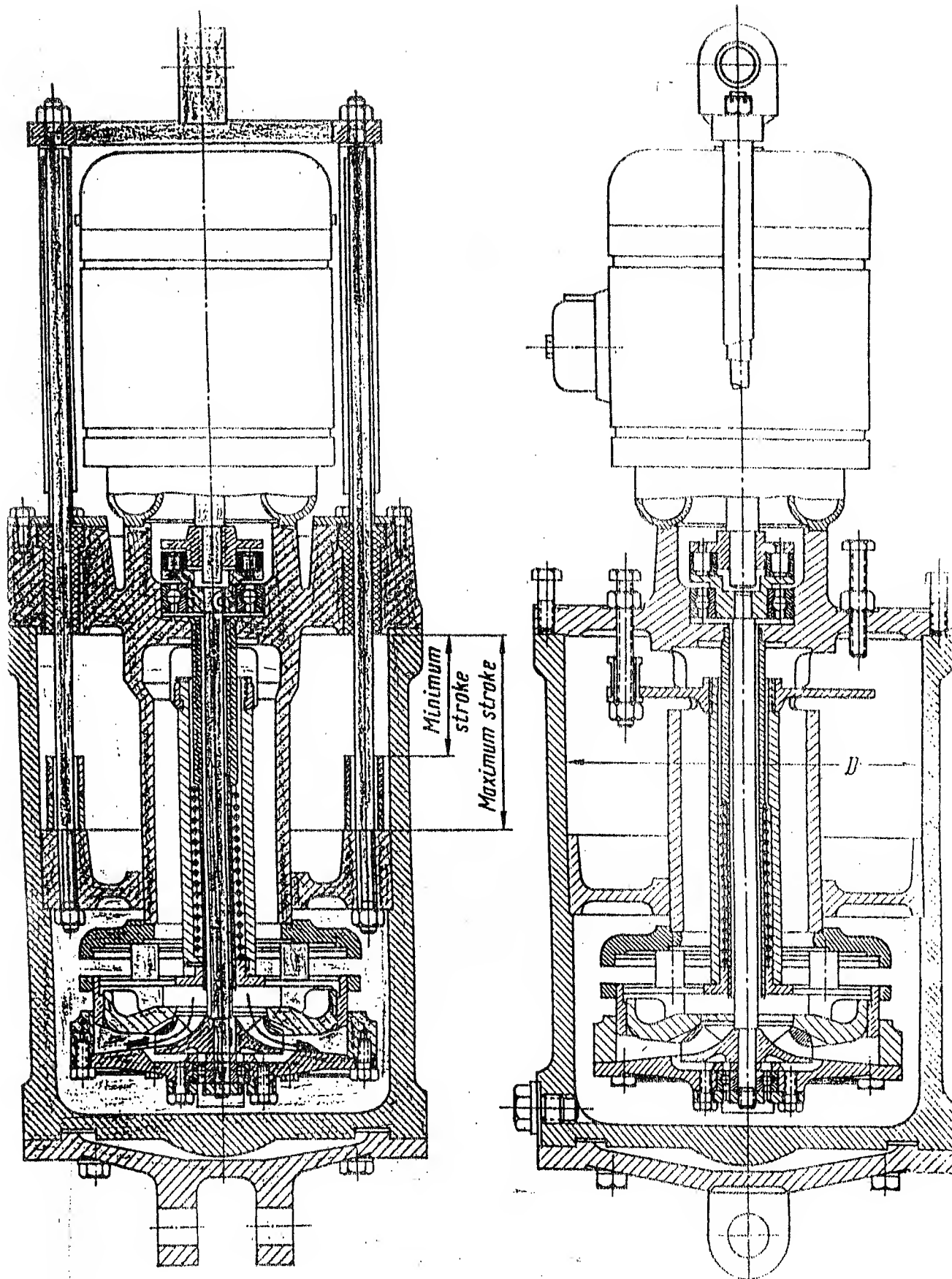
- Drum diam/width 200/100 mm
- Drum peripheral speed = 8 m/s
- Safety margin = 4

§ Required:

If the Electrohydraulic thrustor is the suitable thrustor to disconnect the brake, you are asked to:

- * Work out the forces and reactions on different elements.
- * Make a complete stress analysis for the elements based on the loading diagrams and hence select the suitable materials.
- * Construct the details of the joints necessary to realize the function.
- * What is the max. torque this brake can arrest?





Electrohydraulic thruster
338.

Shoe Brake Projects:

I - Selection of appropriate thruster size:-

- 1- Drum size = \checkmark ; Shoe width = $0.3-0.4$ drum diam. $\checkmark\checkmark$
- 2- Brake shoe angle = 75° \checkmark , Drum size = \checkmark ;
projected length = $\checkmark\checkmark$
- 3- Projected length = \checkmark , width = \checkmark ; area = $\checkmark\checkmark$
- 4- Select lining material (table) \checkmark , allowable
pressure = \checkmark , Service factor = $1.5-2$ \checkmark ;
total reaction = $\checkmark\checkmark$
- 5- Right-hand lever main dimensions \checkmark , reaction
at intermediate point = \checkmark ; tension in tie rod = $\checkmark\checkmark$
- 6- Actuating lever dimensions \checkmark , $P_{actuator} = 0.0$,
tension in tie rod = \checkmark ; $F_{spring} = \checkmark\checkmark$
- 7- Actuator lever dimensions = \checkmark , $F_{spring} = \checkmark$,
 $F_{rod} = 0$; $P_{actuator} \geq \checkmark\checkmark$

Hence Select the actuator.

~ 16

* How to enhance and adjust the calculated values for the available actuator sizes?

Either:

increase the arm length (and hence the stroke)

OR:

reduce projected area by reducing the shoe width (i.e. the reaction decreases, F_{rod} gets less, F_{spring} gets smaller and consequently the actuator force will decrease.)

In general: both approaches could be combined in an adequate way if each alone is not enough.

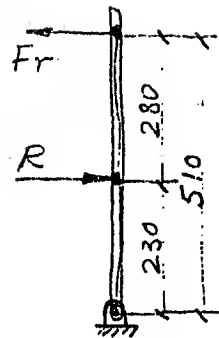
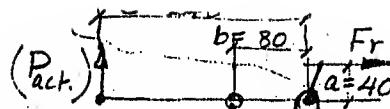
Next step:

Proceed like the solved example to get through the design fixing the material, the selected dimensions in the Elevation View and work out the limit load carrying capacity of this brake arrangement.

$$\phi_{\text{drum}} = 400 \text{ mm}$$

$$\text{shoe width} = 0.3 \rightarrow 0.4 \phi_{\text{drum}} = 120 \rightarrow 160 \text{ mm}$$

$$\text{Projected length of shoe} = 200 \text{ mm}$$



Selection of actuator:

Shoe width	120	140	150	160
Area (mm ²)	24,000	28,000	30,000	32,000
80% Area (mm ²)	19,200	22,400	24,000	25,600
Max. pressure (kPa)		690		
Service factor (≈ 1.5)				
Adopted pressure (kPa)		450		
$R = p \times \text{Area}$ (kN)	8.64	10.1	10.8	11.52
$F_r = (R \times 230) / 510$ (kN)	3.9	4.6	4.9	5.2
$F_s = F_r \times \frac{40}{80}$ (kN)	1.85	2.3	2.45	2.6

$$* \text{Pact. to get } F_r = 0 = P_{\text{act. min}} = \frac{F_s \times 80}{240} \text{ (kN)}$$

$$\text{For } C = 300, b = 80, P_{\text{act. min}} = \frac{F_s \times 80}{300} \text{ (kN)}$$

Select $P_{\text{act. max}} = 600 \text{ N}$, Max stroke of 75 mm

For stroke = 70 mm.:

$$F_{s_{\text{max}}} = 600 \times \frac{300}{80} \text{ kN} = 2.25$$

$$\delta_{\text{spring}} = 70 \times \frac{80}{300} \text{ mm} \approx 19$$

$$F_{s_{\text{max}}} - F_s = \Delta F_s \text{ kN} = 0.4$$

$$K = \text{Spring stiffness} = \frac{\Delta F_s}{\delta_{\text{spring}}} \text{ N/mm} = 21$$

$$\text{Assembly } \delta_{\text{spring}} = \frac{F_s}{K_{\text{spring}}} = \frac{1850}{21} \text{ mm} = 88$$

$$\delta_{\text{total}} = \delta_{\text{spr.}} + \delta_{\text{ass.}} = 107$$

$$n = 20 \text{ eff. coils} = 341$$

Single spring:

$$K = \frac{Gd}{8C^3n}$$

$$\tau = \frac{8PC}{\pi d^2} k$$

$$k = \frac{4C+2}{4C-3}$$

index C	6	7	8
K (N/mm)		21	
δ assembly (mm)		88	
δ working (mm)		19	
n (-)		20	
$d = \frac{8KC^3n}{G} = \frac{8 \times 21 \times 20}{8 \times 10^4} C^3$	9.072	14.406	21.504
$D_o = d(C+1)$	63.5	115.25	193.54
$k = \frac{4C+2}{4C-3} = \frac{26}{21}$	1.238	not suitable for the construction	
total clearance at P_{max} (mm)	23		
δ Free to solid = 88+19+23	130		
$P_{peak} = \delta * K$ (N)	2730		
$\tau = \frac{8PC}{\pi d^2} = \frac{8 \times 2730 \times 1.238}{\pi (9)^2}$	106	N/mm ²	

Material Music wire DIN 17223 Sheet 1, $\tau_{perm} = 675 \text{ N/mm}^2$

Solid length $d \times (n+2)$	≈ 200	mm
Free length = $l_{solid} + \delta$	330	mm
Assembled length = $l_{free} - \delta$	242	mm
D_m	55	mm
D_o	64	mm
D_i	46	mm

Use steel pipe 3" (d/D) $\approx 78/89$ mm.

& a steel pipe $1\frac{1}{4}$ " (33/42 mm) as an inside guide.

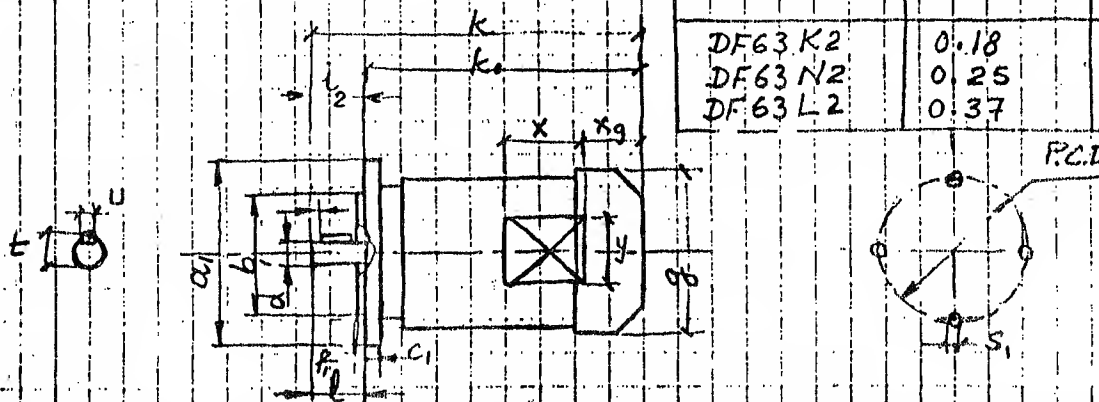
Since: $(3D_o = 3 \times 64 = 192 \text{ mm} < 330 \text{ mm})$

* Experimental Characteristics of Electrohydraulic Thruster

Item	Characteristics	Units	Type of Thruster and Values				
			T-I-55/75		T-II-55/135		
1	Rated force	N	600		1600		
2	Maximum stroke	mm	55	75	55	85	135
3	Duration of max. hoisting	s	0.5	0.73	0.65	1.0	1.5
4	Rated work	N.m	33	48	96	144	224
5	Speed of motor	r.p.m	2840 (3000)				
6	Motor rating	kW	0.144		0.326		
7	Oil pressure	MPa	0.031		0.03		
8	Piston diam.	mm	170		270		
9	Pump delivery	l/s	2.16		4.93		
10	Approximate wt. (dry)	N	400		700		

** Data Sheet for Fraction AC-Squirrel-Cage Motors

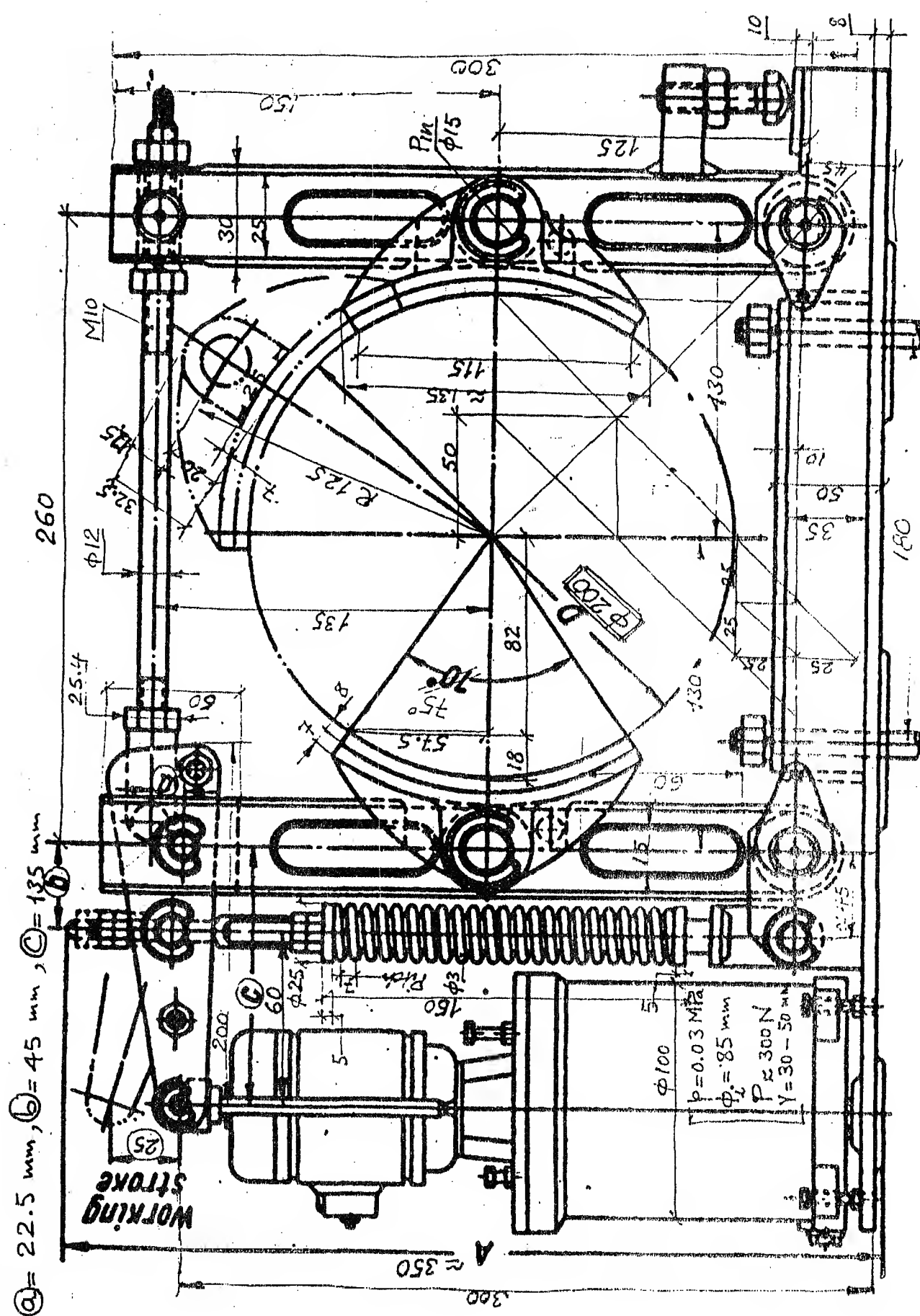
Frame type	Power kW	R.R.M.
DF63 K2	0.18	2720
DF63 N2	0.25	2660
DF63 L2	0.37	2650

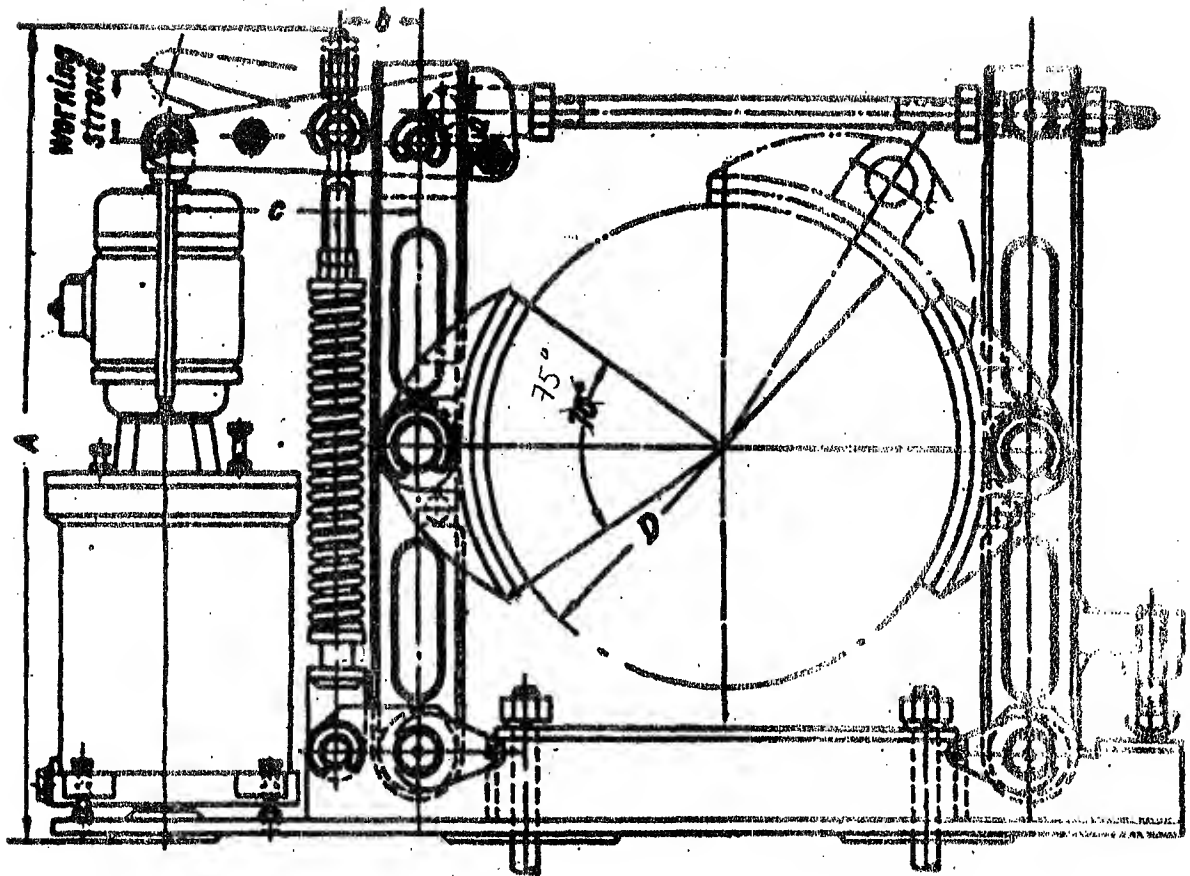


3000 r.p.m.		a_1	b_1	C_1	f_1	l_2	k	k_0	S_1	x_g	$\frac{x}{y}$	$\frac{d}{l}$	$\frac{E}{U}$	$P.C.$
Frame														
DF63...		140	95	9	3	23	215	192	9	54	$\frac{100}{100}$	$\frac{11}{23}$	$\frac{12.6}{4}$	115

* Materials Handling Equipment,
N. RUDENKO, MIR Pub. MOSCOW

** SEW Eurodrive, TUV certified,
Geared Motors Catalogue 1993





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Thrustor : (Electrohydraulic)

$P_{act} = 300 \text{ N} = \text{max. output of the thrustor}$

Stroke = 30 mm (min.), 50 mm (max.)

$P_{act} = 300 \text{ N}$

Stroke = 30 mm

$F_r = \text{tie rod force}$

$F_s = \text{Spring force when}$

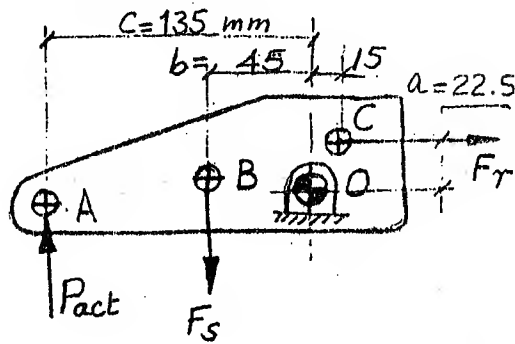
$P_{act} = 0$

$F_s = \text{max. spring force at the end of the lever stroke.}$

When the brake

is just released, $F_r = 0$

Moment about O = 0.0



$$P_{act} \times C = F_s \times b \quad \therefore F_s = \frac{300 \times 135}{45} = 900 \text{ N}$$

$$F_s = 900 \text{ N}$$

Since P_{act} is the max. thrustor force, then the $F_s = 900 \text{ N}$ is the spring force when fully deflected at the end of thrustor stroke.

Spring dimensional chs:

$$n_{eff} = 20$$

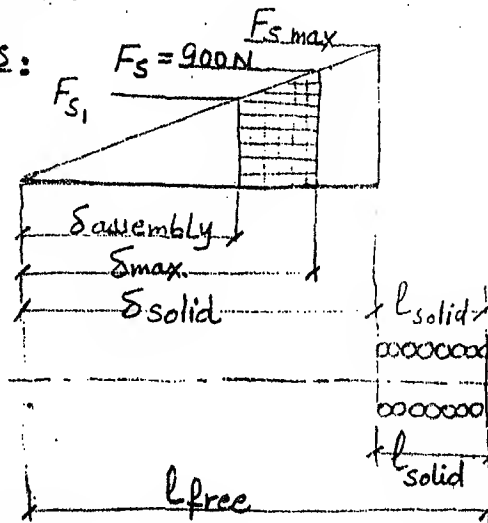
$$D_o = 25 \text{ mm}$$

$$d = 3 \text{ mm}$$

$$\therefore D_m = 22 \text{ mm}$$

$$C = \frac{22}{3} = 7.33$$

$$k = \frac{4C + 2}{4C - 3} = \frac{31.32}{26.32} = 1.2$$



Spring assume
dims.:

$$n_{eff} = 20$$

$$D_o = 25 \text{ mm}$$

$$d = 3 \text{ mm}$$

Square ground ends.

$$k = 1.2$$

Spring stiffness = K

$$K = \frac{G d}{8 C^3 n} = \frac{8 \times 10^4 \times 3}{8 (7.33)^3 \times 20} = 3.8 \text{ N/mm}$$

$$K = 3.8 \text{ N/mm}$$

$$\delta_{work} = \delta_{max} - \delta_{ass} = \text{thrustor stroke} \times \frac{45}{135} = 10 \text{ mm}$$

$$\delta_{work} = 10 \text{ mm}$$

$$\therefore F_{s1} = 900 - (3.8 \times 10) = 862 \text{ N}$$

$$l_{solid} = n_{eff} \times d + 2d \rightarrow \text{Square ground ends}$$

$$\therefore l_{\text{solid}} = 22 \times 3 = 66 \text{ mm}$$

$$\delta_{\text{ass}} = F_{S_1} / K = 862 / 3.8 = 227 \text{ mm}$$

Since $l_{\text{ass}} = 150 \text{ mm}$ and, $\text{pitch}_{\text{ass}} = 7$

$$\text{then, } l_{\text{free}} = 150 + 227 = 377 \text{ mm} = n \times \text{pitch} + 2d$$

$$\text{i.e. } \text{pitch}_{\text{free}} = (377 - 6) / 20 = \frac{371}{20} = 18.5 \text{ mm}$$

$$\delta \text{ to solid spring} = l_{\text{free}} - l_{\text{solid}} = 377 - 66 \\ = 311 \text{ mm (not practical)}$$

$$\tau \geq \frac{8 P_{\text{max}} C}{\pi d^2} k \quad \text{where}$$

$$F_{S \text{ max}} = P_{\text{max}} = 311 \times 3.8 = 1182 \text{ N}$$

$$\tau_{\text{perm}} \geq \frac{8 \times 1182 \times 7.33 \times 1.2}{\pi (9)} \geq 2941 \text{ N/mm}^2$$

$$\tau_{\text{work}} = 2941 \times \frac{900}{1132} = 2240 \text{ N/mm}^2 \rightarrow \text{Too high}$$

Three trials for $d = 3.5, 4, 4.5$

Item	3	3.5	4	4.5
$C = \frac{25-d}{d}$	7.33	6.14	5.25	4.56
$k = \frac{4C+d}{4C-3}$	1.2	1.23	1.28	1.33
$K = \frac{G d}{8 C^3 n} = \frac{10^4 d}{20 C^3} \text{ [N/mm]}$	3.8	7.57	13.8	23.7
$l_{\text{solid}} = (n+2)d \text{ [mm]}$	66	77	88	99
$F_{S_1} = 900 - 10K \text{ [N]}$	862	824	762	663
$\delta_{\text{ass}} = F_{S_1} / K \text{ [mm]}$	227	109	55	28
$l_{\text{free}} = 150 + \delta_{\text{ass}} \text{ [mm]}$	377	259	205	178
$\delta_{\text{solid}} = l_{\text{free}} - l_{\text{solid}}$	311	182	113	79
$P_{\text{max}} = F_{S \text{ max}} = K \delta_{\text{solid}} \text{ [N]}$	1182	1378	1560	1872
$\tau = \frac{8 P_{\text{max}} C}{\pi d^2} \cdot k$	2941	2163	1668	1427
$\tau_{\text{working}} \text{ [N/mm}^2\text{]}$	2240	1413	962	686
Material No	347			6

τ_{perm} is too high.

Appropriate Spd

$$K = 23.7 \text{ N/m}$$

Assembly force
663 N

$$\delta_{\text{ass}} = 28 \text{ mm}$$

$$l_{\text{free}} = 178 \text{ mm}$$

$\tau_w < 860 \text{ N/mm}$
oil Tempered
Cr. Si

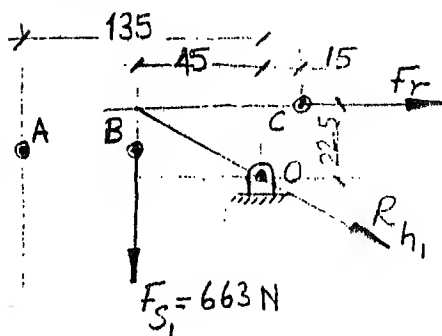
A) Force Analysis:

D-3/19

Forces in holding brake elements: (internal forces)

1) Actuating lever:

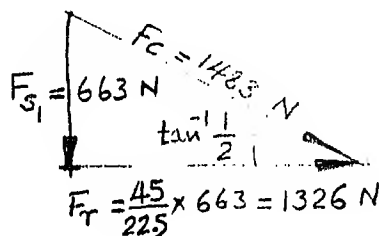
Based on $F_{S1} = 663 \text{ N}$,
 $\therefore F_T = 1326 \text{ N}$ along the
 top tie rod



$$F_{S1} = 663 \text{ N}$$

$$F_T = 1326 \text{ N}$$

Force Polygon.



2) Shoes Carrying Levers:

Mts @ O_1 : [L.H. Lever]

$$125 * R = (125 + 120) * 1326$$

$$\therefore R_L = 2600 \text{ N}$$

μ = Coefficient of friction

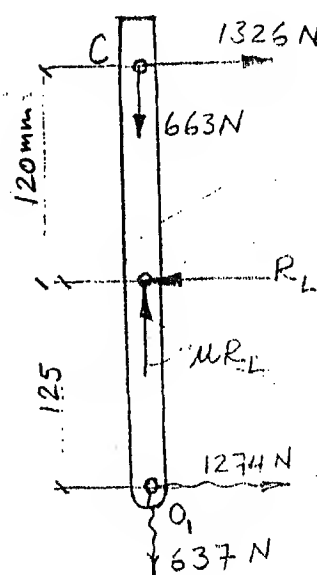
$$\therefore \mu = 0.5$$

$$\therefore \mu R_L = 1300 \text{ N}$$

Reaction at O_1 :

$$R_{O1x} = 2600 - 1326 = 1274 \text{ N}$$

$$R_{O1y} = 1300 - 663 = 637 \text{ N}$$



Mts @ O_2 : [R.H. Lever]

$$R * 125 = 1326 * 260$$

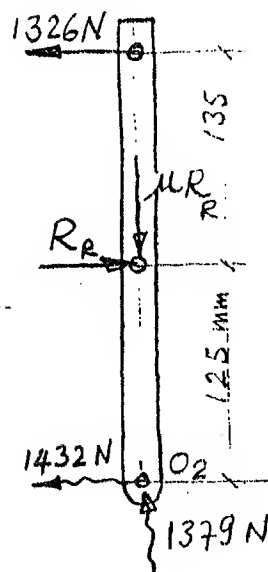
$$\therefore R_R = 2758 \text{ N}$$

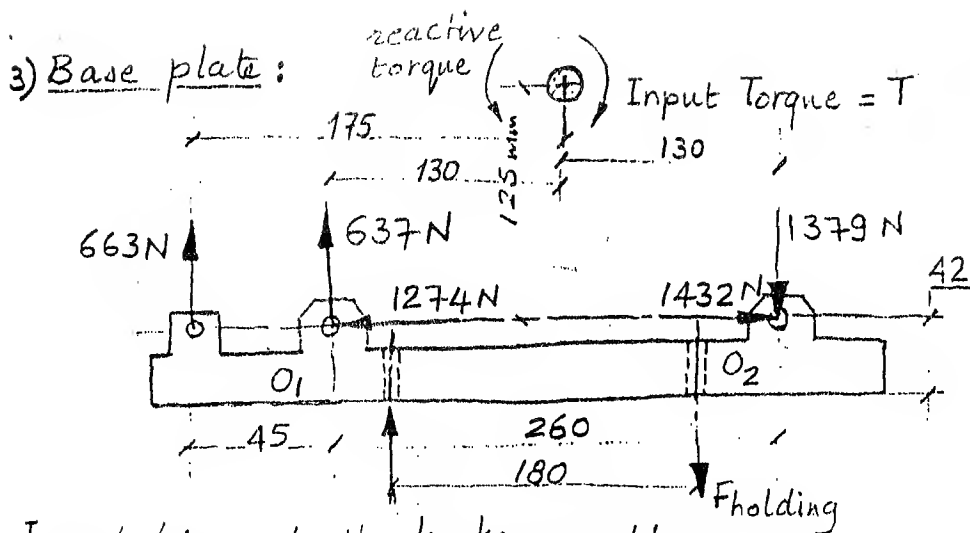
$$\mu R_R = 1379 \text{ N}$$

Reactions at O_2 :

$$R_{O2x} = 2758 - 1326 = 1432 \text{ N}$$

$$R_{O2y} = 1379 \text{ N}$$





Input torque to the brake assembly:

$$T_{\text{work}} = -[(1432 - 1274) \times 125] + [(1379 + 637) \times 130] + [663 \times 175]$$

$$= -19750 + 262080 + 110775 = 353105 \text{ N}\cdot\text{mm}$$

$$= 0.353 \text{ kN}\cdot\text{m}$$

$$T_{\text{work}} = 0.353 \text{ kN}\cdot\text{m}$$

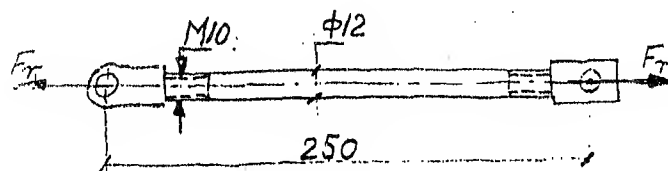
4) Holding down bolts: (Two bolts)

$$F_{\text{holding}} = \frac{353105}{180} = 1962 \text{ N}$$

Holding down force in each bolt = 1962 N

5) Tie rod (coupler):

$$F_T = 1326 \text{ N}$$



$$F_T = 1326 \text{ N}$$

The rod construction details should allow for fine adjustment of brake clearance.

Stresses in different members:

1) Right-hand Lever:

Material: C15

$$\sigma_y = 300 \text{ N/mm}^2$$

$$B.M. = 2758 \times 125 = 179 \times 10^3 \text{ N}\cdot\text{mm}$$

Since the element is used in lifting equipment, the margin of safety should be as high as 4

$$\therefore \sigma_{\text{design}} = \frac{300}{4} = 75 \text{ N/mm}^2$$

$$\sigma_{\text{design}} = M/Z$$

$$\therefore Z = \frac{M}{\sigma_{\text{design}}} = \frac{179 \times 10^3}{75} = 2387 \text{ mm}^3$$

$$Y = \frac{25}{2} = 12.5 \text{ mm}$$

$$I = ZY = 12.5 \times 2387 \approx 30,000 \text{ mm}^4$$

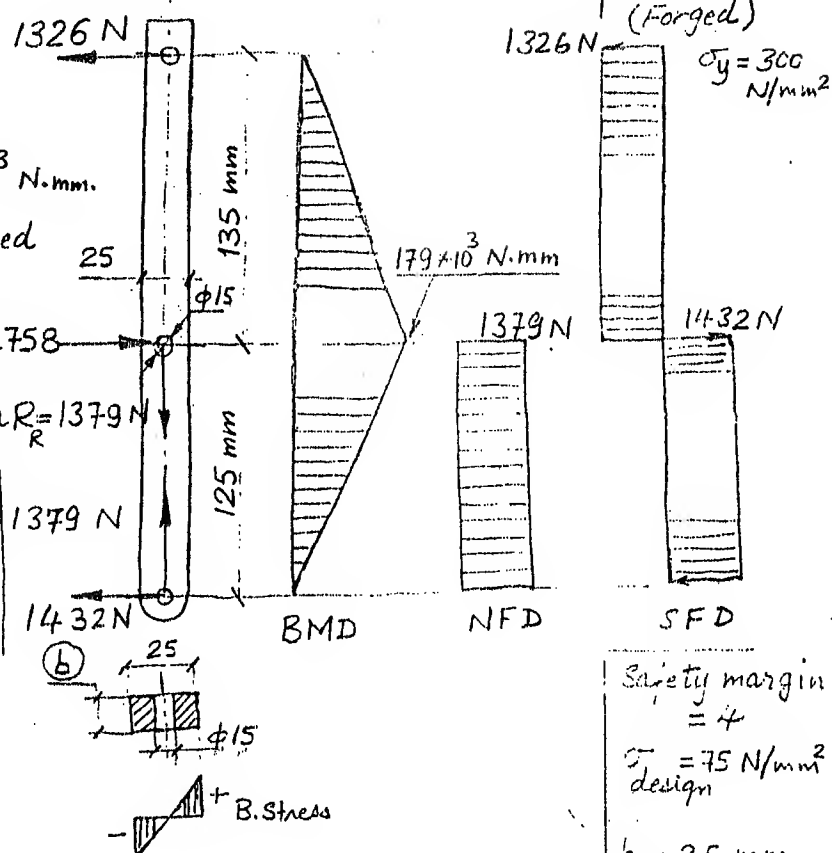
$$I = \frac{b}{12} (25^3 - 15^3) = b(12250)/12 = 30,000 \text{ mm}^4$$

$$\therefore b = 29 \text{ mm taken } 30 \text{ mm}$$

Note:

The normal force of 1379 N results in compression which reduces the tensile stress

The shear stress has a max. value at the neutral axis of the section where the bending stress is zero.



D-5/19

Material:
St C15
(Forged)

$$\sigma_y = 300 \text{ N/mm}^2$$

Safety margin
= 4

$$\sigma = 75 \text{ N/mm}^2 \text{ design}$$

$$h = 25 \text{ mm}$$

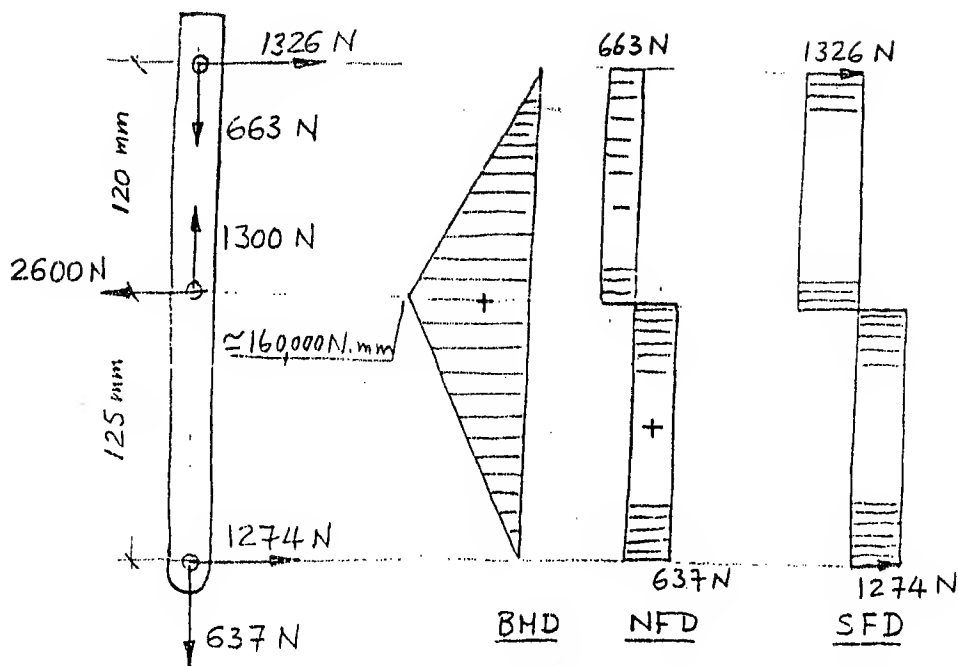
$$Y = 12.5 \text{ mm}$$

$$I = 30,000 \text{ mm}^4$$

$$b = 30 \text{ mm}$$

2) Left-hand lever :

D-6/19



$$B.M. = 160,000 \text{ N}\cdot\text{mm}$$

$$N.F = 637 \text{ N}$$

$$I = \frac{b}{12} (25^3 - 15^3) = \frac{b}{12} (12250) \text{ mm}^4$$

$$A = b(25 - 15) = b(10) \text{ mm}^2$$

$$\sigma_{t_{total}} = \sigma_t + \sigma_b$$

$$= \frac{637}{10b} + \frac{160000 \times 12.5 \times 12}{12250 b}$$

$$= \frac{63.7}{b} + \frac{1959}{b} = \frac{2023}{b}$$

$$\sigma_{t_{design}} = \frac{300}{4} = 75 \text{ N/mm}^2$$

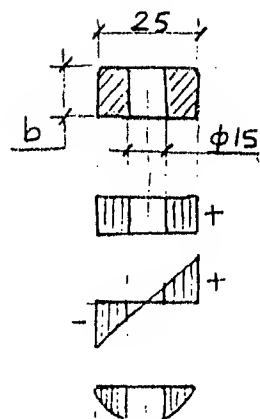
$$\therefore \frac{2023}{b} = 75 \text{ i.e. } b = 27 \text{ mm}$$

Unification of as forged levers is a required.

$\therefore b$ is taken 30 mm

$$\therefore \sigma_{t_{total \text{ working}}} = \frac{2023}{30} = 67.4 \text{ N/mm}^2$$

$$\therefore \text{Working safety factor} = \frac{300}{67.4} = 4.45$$



$$\sigma_t = 67.4 \text{ N/mm}^2$$

$$\text{Working S.F.} = 4.45$$

3) Actuating Lever:

$$P_{act} = 300 \text{ N}$$

$$F_s = 900 \text{ N}$$

$$R_o = 600 \text{ N}$$

Lever thickness = 12 mm

Material: St C15

$$\sigma_{yt} = 300 \text{ N/mm}^2$$

$$SF = 4 \text{ i.e. } \sigma_{t \text{ design}} = 75 \text{ N/mm}^2$$

$$\sigma_{t \text{ design}} = \frac{M}{Z}$$

$$\therefore Z \geq \frac{27000}{75} \geq 360 \text{ mm}^3$$

$$Z = \leq \frac{bh^3}{6} = \frac{10h^3}{6} - \frac{12d^3}{6}, \quad d = 12 \text{ mm}$$

$$\therefore 360 = 1.67(h^3) - 2(12^3) = 1.67h^3 - 288$$

$$h = \sqrt[3]{(360 + 288)/1.67} = 20 \text{ mm}$$

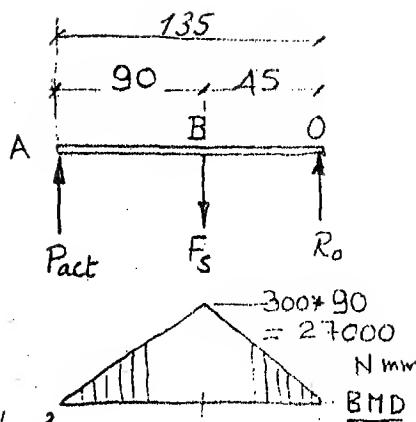
take $h \geq 24 \text{ mm}$ with a hole $\phi 12$ about the neutral axis of the section in plane at B.

D-7/19

Material:

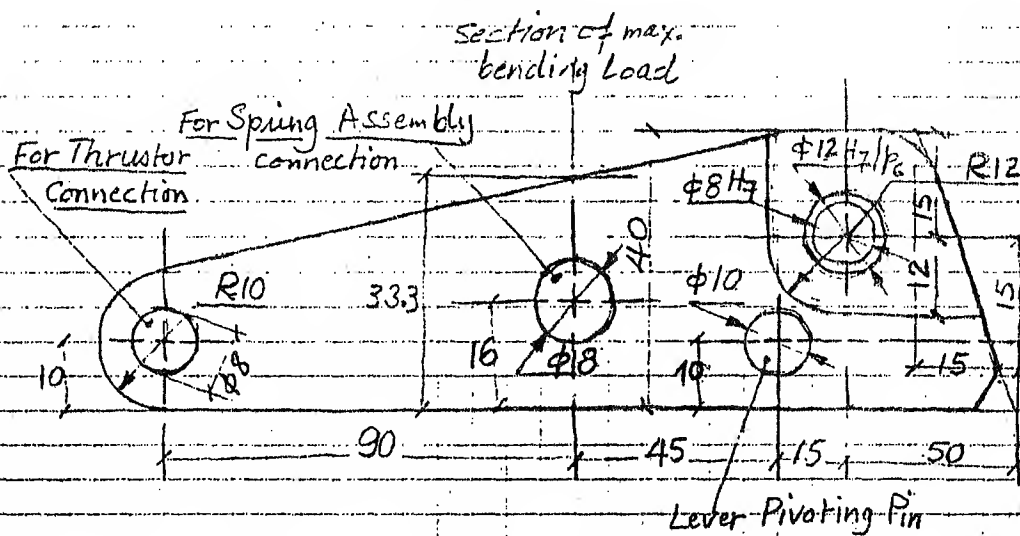
C15

$$\sigma_{yt} = 300 \text{ N/mm}^2$$



$$Z \geq 360 \text{ mm}^3$$

$$\therefore h \geq 20 \text{ mm}$$

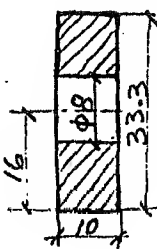


$$Z_{\text{actual}} = \frac{10(33.3)^3}{6} - \frac{10(8)^3}{6}$$

$$= 1740 \text{ mm}^3$$

$$\sigma_{\text{working}} = \frac{27000}{1740} = 15.5 \text{ N/mm}^2$$

Working margin of safety = 19



$$Z_{\text{act}} = 1740 \text{ mm}^3$$

Working margin of safety = 19
(rigid enough)

352.

- Bending stress on hinge pin:

$$d = \text{Pin diam.} = 8 \text{ mm}$$

$$Z = \frac{\pi d^3}{32} = 50.3 \text{ mm}^3$$

$$M_b = 5304 \text{ N}\cdot\text{mm}$$

$$\sigma_b = \frac{5304}{50.3} = 105.5 \text{ N/mm}^2$$

$$\sigma_y \geq \sigma_b * \text{margin of safety}$$

$$\text{margin of safety} = 4$$

$$\therefore \sigma_y \geq 105.5 * 4 = 422 \text{ N/mm}^2$$

Pin material is st C60, $\sigma_y = 490 \text{ N/mm}^2$

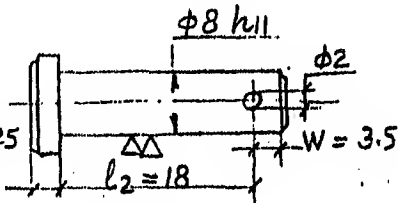
Note:

Calculating the pin in this way is an extreme case.

For St C45, $\sigma_y = 400 \text{ N/mm}^2$ is more common than St C60 but the yield stress (min. guaranteed) is 400 N/mm^2 which is less than the required value by $22 \text{ N/mm}^2 = 5.5\% \sigma_y$ and can be used instead of St C60.

\therefore Hinge pin material is St C45

- DIN 1444, Form B
- Ordinary washer DIN 125
 $D = 18, d = 9.5$
and $t = 2 \text{ mm}$
- Split Pin:
DIN 935, $d \times l = 2 \times 18 \text{ mm}$



- Bearing pressure on hinge pin: [L.H. Lever]

$$\text{Pin / actuating Lever, Bearing area} = l d = 8 * 8 = 64 \text{ mm}^2$$

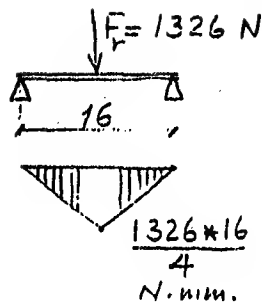
$$\text{Bearing pressure} = \frac{1326}{64} = 20.7 \text{ N/mm}^2$$

$$\text{The allowable bearing pressure for Br/st} = 17 \text{ N/mm}^2$$

i.e. Using a bronze bush is a must with limited, but reasonable, life since the motion is limited.

$$d = 8 \text{ mm}$$

$$l = 16 \text{ mm}$$



$$\sigma_{b \text{ working}} = 105.5 \text{ N/mm}^2$$

Material:

st C60

$$\sigma_y = 490 \text{ N/mm}^2$$

and replaced by

st C45

$$\sigma_y = 400 \text{ N/mm}^2$$

Std. hinge Pin,

DIN 1444

Form B

Washer, DIN 125,

$D = 18 \text{ mm}$

Split Pin, DIN 935

$\phi 2 \times 18$

Bearing Pressure

$$\approx 21 \text{ N/mm}^2$$

$$= 21 \text{ MPa}$$

St/Bronz bush

4) Tie Rod : (Coupler)

Thread size M10 x 1.25, core area = 61.2 mm²

The min. section is the core area.

$$\text{Force} = F_r = 1326 \text{ N}$$

$$\text{S.F.} = 4$$

$$\text{Actual stress} = \frac{1326}{61.2} = 21.7 \text{ N/mm}^2$$

∴ The material should have $\sigma_y \geq 21.7 \times 4 \geq 87 \text{ N/mm}^2$

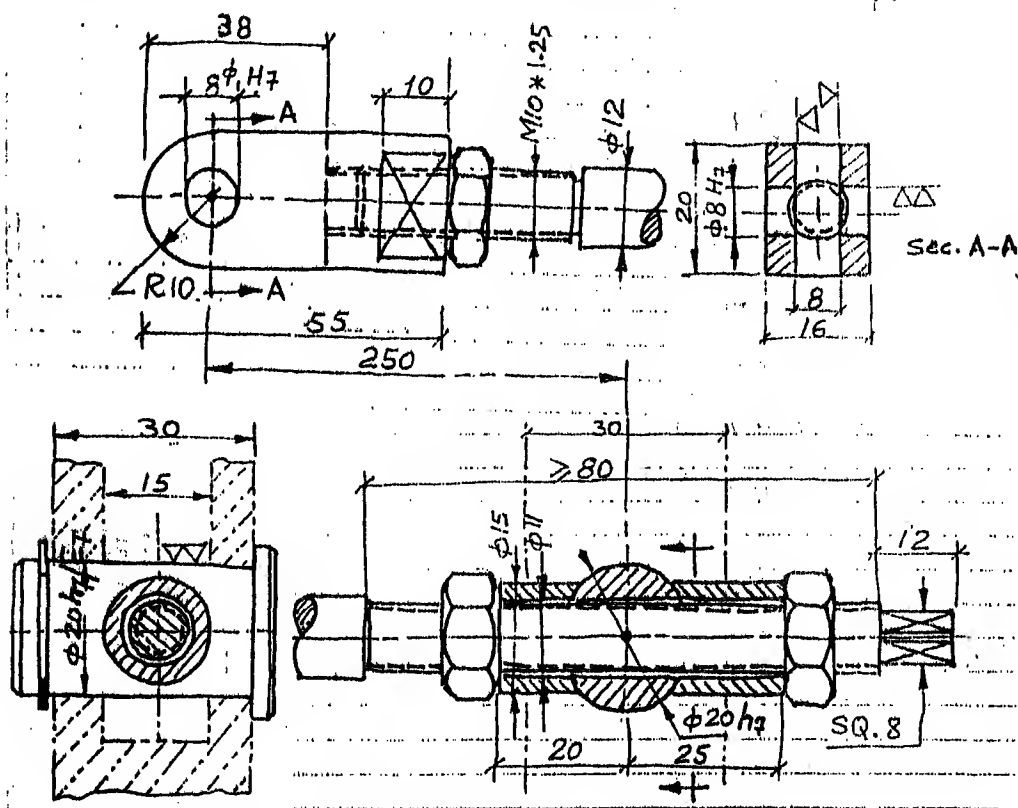
For st 34, $\sigma_y = 190 \text{ N/mm}^2$

∴ The coupler material is st 34

M10 x 1.25 mm
Core Area = 61.2 mm²

$$\sigma_{\text{working}} = 21.7 \text{ N/mm}^2$$

Material: st 34,
 $\sigma_y = 190 \text{ N/mm}^2$



- Tensile stress on yoke eye:

$$\text{area of section A-A} = 4(4 \times 6) = 96 \text{ mm}^2$$

$$\text{tensile stress} = \frac{1326}{96} = 13.8 \text{ N/mm}^2$$

Suitable material is st 34, $\sigma_y = 190 \text{ N/mm}^2$

$$\sigma_{\text{working}} \approx 14 \text{ N/mm}^2$$

Material: st 34
 $\sigma_y = 190 \text{ N/mm}^2$

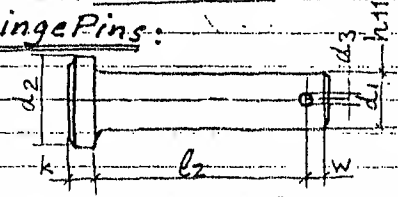
Pipes: (ASA Stds.)

Nominal	d''	D''	T.P.I./ L_1''
1	1.049 (26.6)	1.315 (33.4)	$11\frac{1}{2}/0.4$ (9)
$1\frac{1}{4}$	1.380 (33.02)	1.66 (42.16)	$11\frac{1}{2}/0.42$ (10)
$1\frac{1}{2}$	1.610 (40.9)	1.900 (48.26)	$11\frac{1}{2}/0.42$ (11)
2	2.067 (52.5)	2.375 (60.33)	$11\frac{1}{2}/0.43$ (12)
$2\frac{1}{2}$	2.469 (62.71)	2.875 (73.03)	$8/0.682$ (18)
3	3.068 (77.93)	3.5 (88.9)	$8/0.766$ (20)

T.P.I. Threads per Inch.

L_1'' Length of engaged threads.

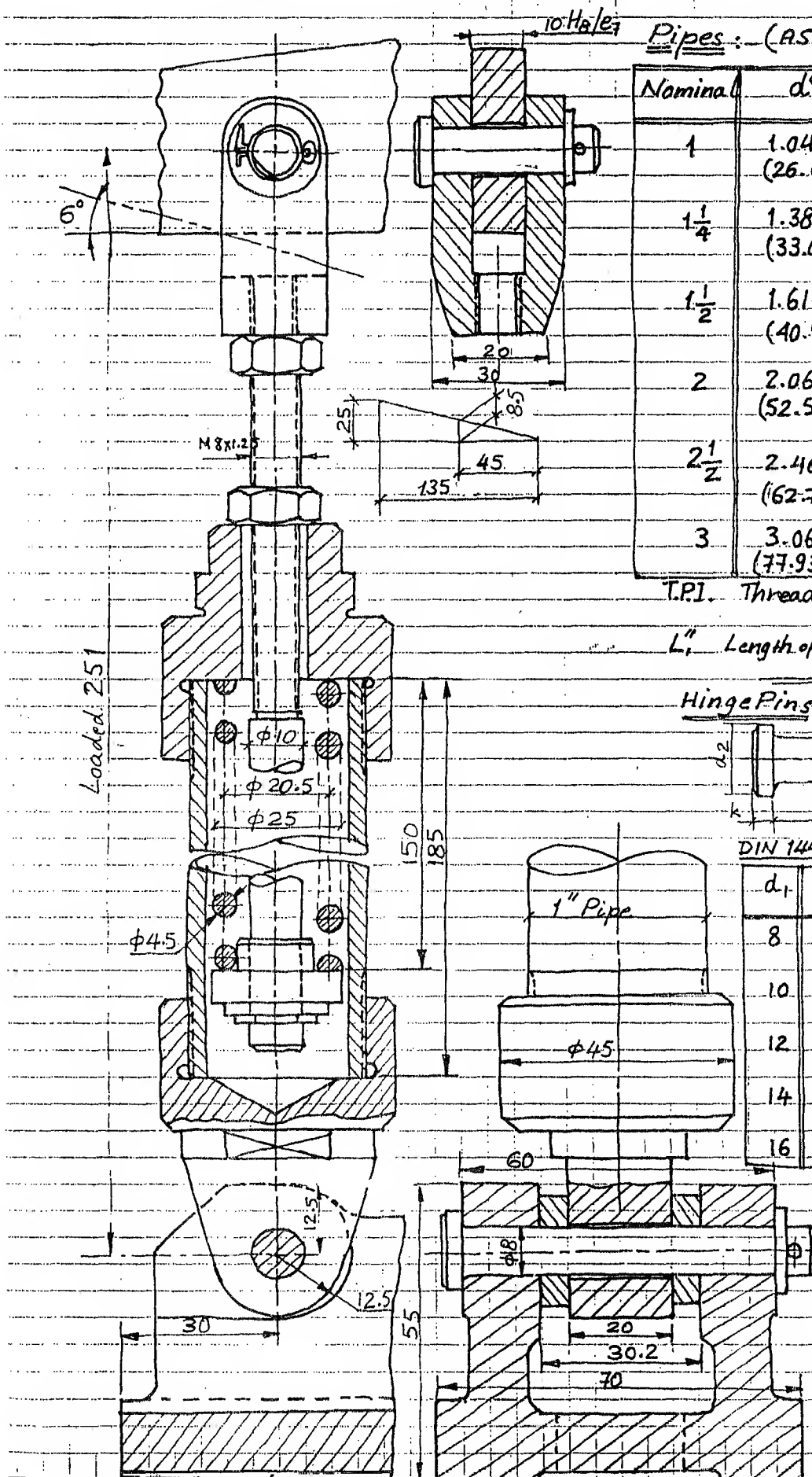
Hinge Pins:

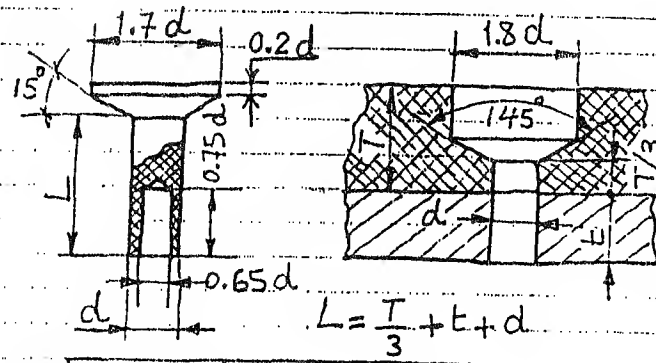


DIN 1444, Form B, Mar. 1974.

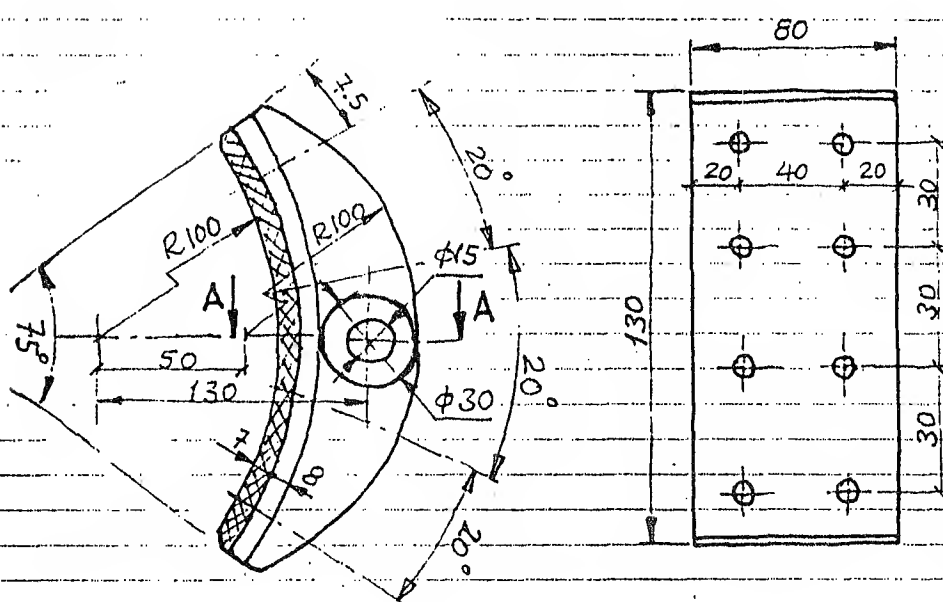
d_1	d_2	d_3	w	k
8	14	2	3.5	3
10	18	3.2	4.5	4
12	20	3.2	5.5	4
14	22	4	6	4
16	25	4	6	4.5

$l_2 = 18, 20, (22), 25, (28), 30, \dots, 100$
5 to 5





Code	D*	E	F	G	H	J
d (mm) metric	2.5	3	3.5	4	4.5	5
d (inch) imperial	3/32	1/8	9/64	5/32	11/64	3/16
* Solid, Flat head rivet						
max. Area/rivet cm ²	4	5.5	7.5	9.5	12	15
T (mm)	5	7	10	11	12	13



$$T = 7 \text{ mm}$$

$$\text{Number of rivets} = \frac{\text{lining area}}{\text{area per rivet}} = \frac{13 \times 8}{15} \approx 8 \text{ rivets}$$

$$L = \frac{T}{3} + t + d = \frac{7}{3} + 8 + 5 = 15 \text{ mm}$$

$$\begin{aligned} \text{No of rivets} &= 8 \text{ rivets} \\ \phi 5, L &= 15 \text{ mm} \end{aligned}$$

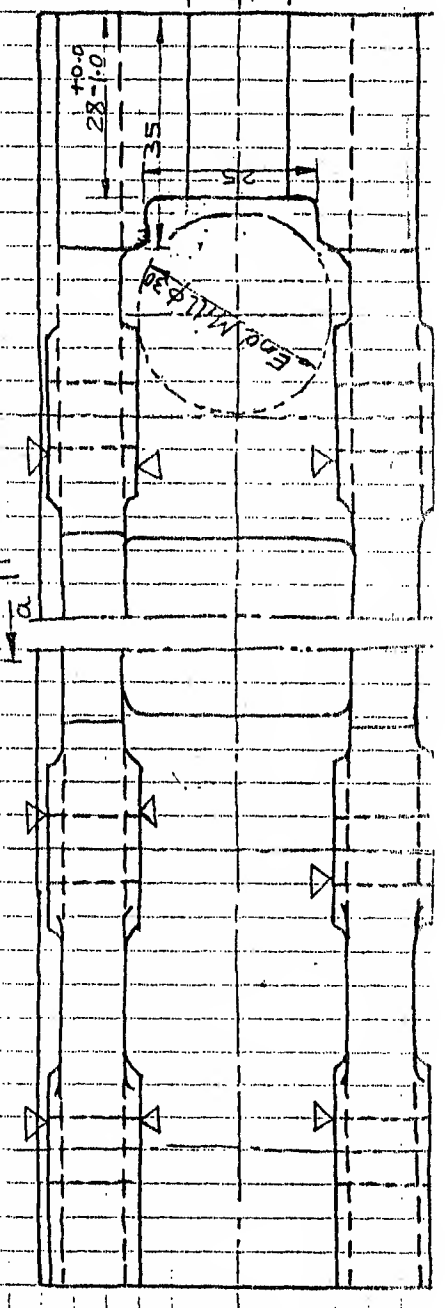
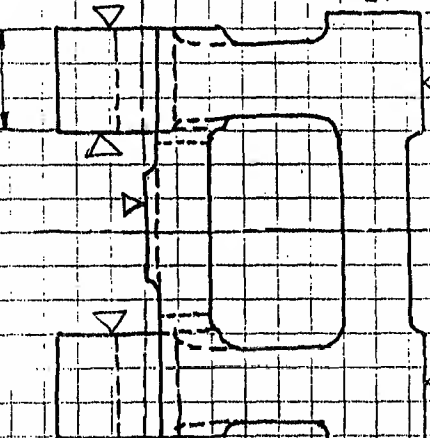
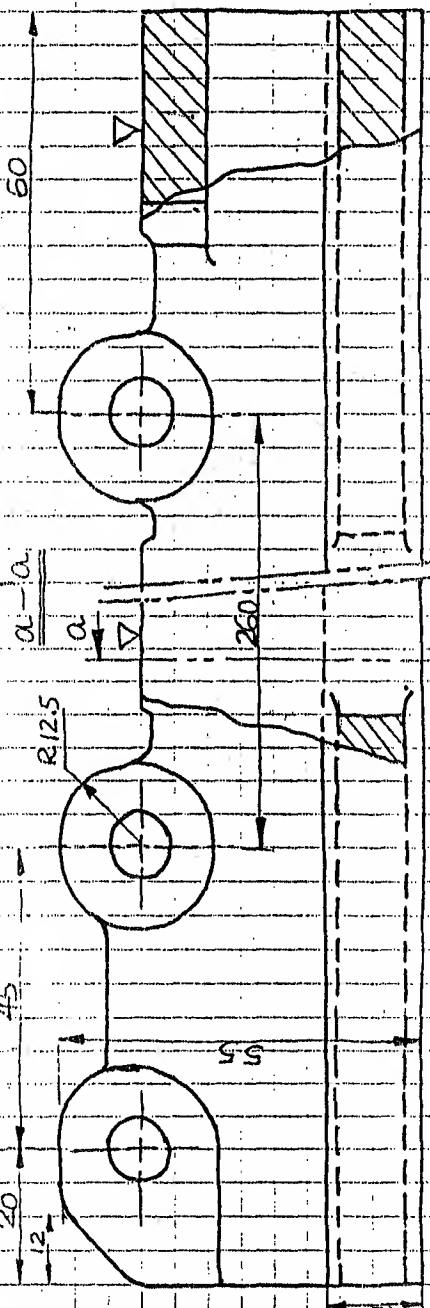
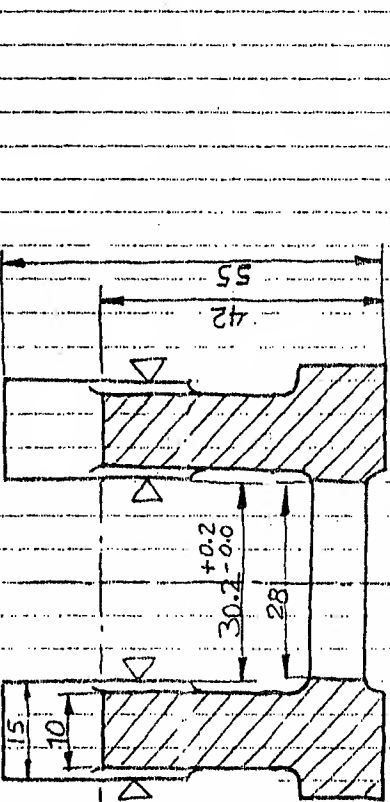
Approximately:

$$(For sec. a-a) I = 2 \left[\frac{10 \cdot (42)^3}{12} \right] = 12.35 \text{ cm}^4$$

$$y = 2.1 \text{ cm}$$

$$Z_b = \frac{I}{y} = 5.88 \text{ cm}^3$$

$$Area = 8.4 \text{ cm}^2$$



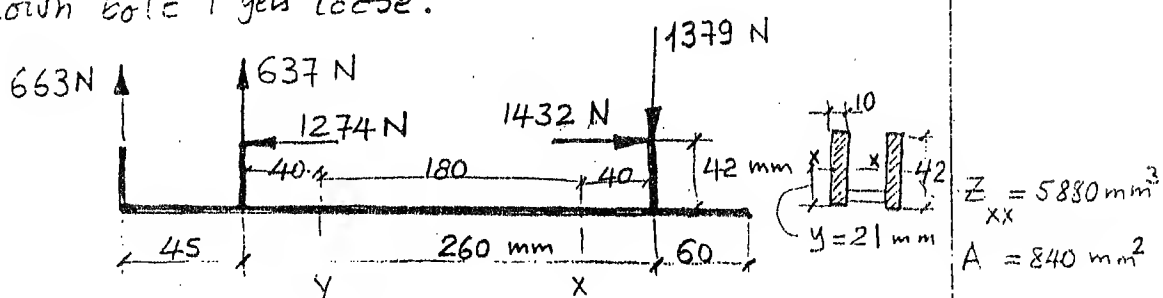
6) R.H. (and L.H) lever / base hinge pin connection:-

$$\text{Resultant force} = \sqrt{(1432)^2 + (1379)^2} = 1988 \text{ N}$$

This force is less than $R = 3084 \text{ N}$ which is the resultant force on the hinge pin of the brake shoe. For the sake of interchangeability, the same pin of the brake shoe; in size and material, is used with a higher working margin of safety.

7) Brake base:

The loading diagram when the holding down bolt Y gets loose:



$$\sum M_{ts} @ X = 0$$

$$M = [663 * (45 + 40 + 180)] + [637 * (40 + 180)] - [1274 * 42]$$

$$= 175,695 + 140,140 - 53,508 = 262,327 \text{ N.mm.}$$

Stressed section properties:

$$Z = 5880 \text{ mm}^3, A = 840 \text{ mm}^2$$

$$\sigma_{\text{norm.}} = \frac{M}{Z} + \frac{F}{A}$$

$$= \frac{262327}{5880} + \frac{1274}{840} = 44.6 + 1.5 = 46 \text{ N/mm}^2$$

$$\sigma = 46 \text{ N/mm}^2$$

max working

$$\sigma_y \geq 46 * \text{margin of safety} \geq 46 * 4 \geq 184 \text{ N/mm}^2$$

The suitable material is cast steel GS 45, $\sigma_y = 220 \text{ N/mm}^2$

Material:

GS 45

$$\sigma_y = 220 \text{ N/mm}^2$$

$$\therefore \text{Working margin of safety} = \frac{220}{46} = 4.78$$

8) Holding down bolts:

Holding down force = 1962 N/bolt

Select the size and look for the suitable material.

"The bolt should be tightened to generate an axial load in excess of the force of 1962 N by about 150% that the other bolt will have $P_T = 50\% P_a$ "

Axial load on the bolt ≈ 3000 NBolt size is M14*1.5, core area = 125 mm²

$$\sigma_t = \text{Normal stress} = \frac{5000}{125} = 40 \text{ N/mm}^2$$

This is the working stress on the highly stressed bolt.

Calculation of the assembly stress

$$T = T_1 + T_2 = P_a \left[\frac{d_m (\tan \alpha + \mu')}{2(1 - \mu' \tan \alpha)} + \mu_s \frac{d_{ms}}{2} \right]$$

$$d_i = \sqrt{\frac{4 \times 125}{\pi}} = 12.6 \text{ mm}$$

$$d_m = \frac{14 + 12.6}{2} = 13.3 \text{ mm}$$

$$\tan \alpha = \frac{p}{\pi d_m} = \frac{1.5}{\pi(13.3)} = 0.072$$

$$\mu_{st/st} = 0.25, \mu'_{st/st} = \frac{0.25}{\cos 30^\circ} = 0.28$$

$$d_{ms} = \frac{22 + 16}{2} = 19 \text{ mm}$$

$$\begin{aligned} \therefore T &= 5000 \left[\frac{13.3}{2} \left(\frac{0.072 + 0.28}{1 - 0.072 \times 0.28} \right) + 0.25 \left(\frac{19}{2} \right) \right] \\ &= 5000 [2.341 + 2.375] = 23580 \text{ N.mm} \end{aligned}$$

$$\tau = \frac{16 \times 5000 \times 2.341}{\pi (12.6)^3} = 29.8 \text{ N/mm}^2$$

$$\tau_{comb} = \frac{1}{2} \sqrt{(40)^2 + 4(29.8)^2} = 35.9 \text{ N/mm}^2$$

$$\tau_y \geq 35.9 \times 1.3 \geq 47 \text{ N/mm}^2$$

$$\sigma_y \geq \tau_y / 0.6 \geq 78 \text{ N/mm}^2$$

$$\sigma_{y_{8.8}} = 64 \text{ N/mm}^2, \sigma_{y_{9.8}} = 72 \text{ N/mm}^2$$

st high grade 10.9 is selected, $\sigma_{y_{10.9}} = 90 \text{ N/mm}^2$

$$\text{Working margin of safety} = \frac{90}{40} = 2.25, \text{Margin}_{\text{assembly}} = \frac{90}{72} = 1.25$$

High Tensile Bolt

M14*1.5

Core Area = 125 mm²

$$\sigma_t = 40 \text{ N/mm}^2$$

Tightening Torque

$$T = 23.6 \text{ N.m.}$$

$$\tau \approx 30 \text{ N/mm}^2$$

$$\tau_{comb} \approx 36 \text{ N/mm}^2$$

$$\sigma_y \geq 78 \text{ N/mm}^2$$

Material

St. Grade 10.9

Margins:

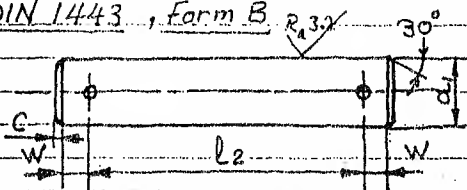
Working = 2.25

Assembly = 1.25

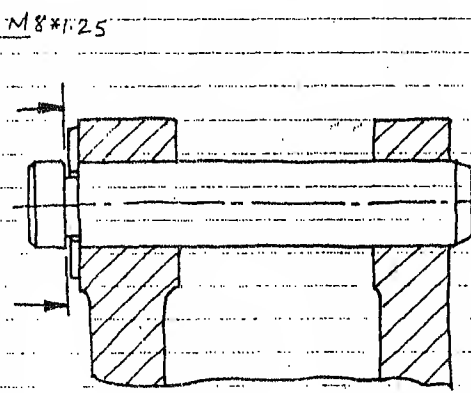
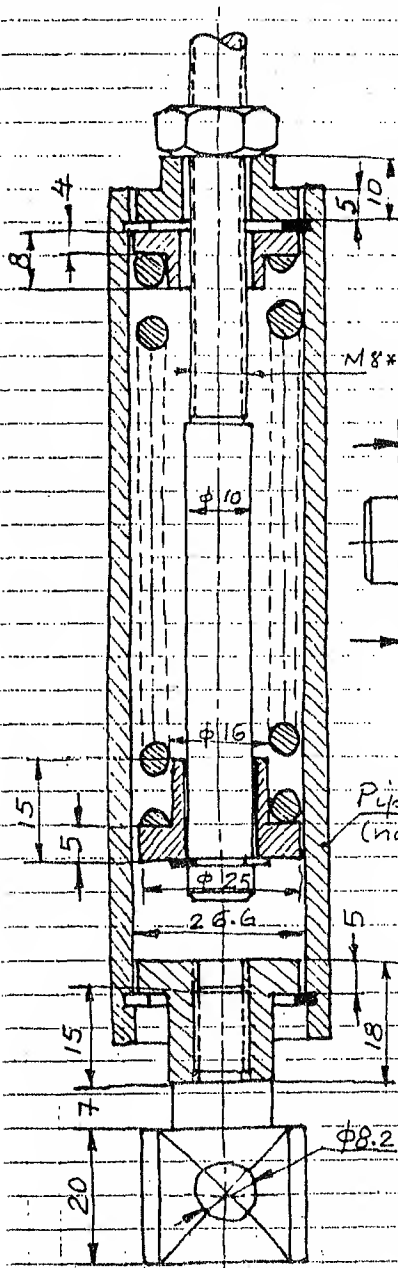
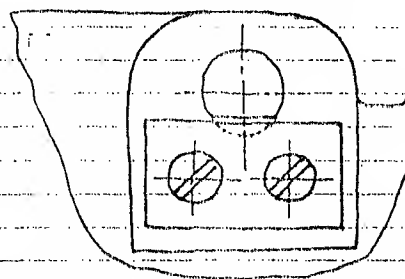
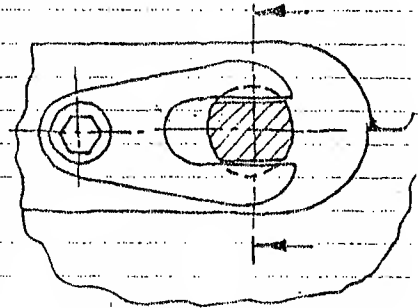
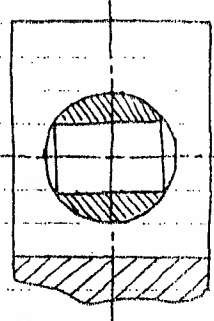
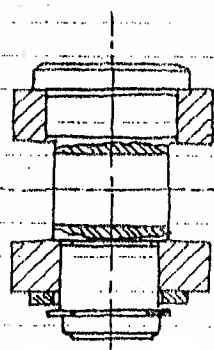
(The same as DIN EN 22340)

DIN 1443, Form B

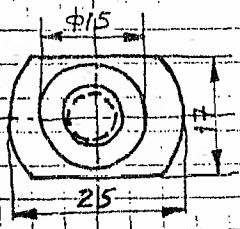
11-19/19



Form A: The same but without split pin hole, length $l_1 = l_2 + 2W$



Pipe 1" (nominal)



Some alternative joints and assemblies.